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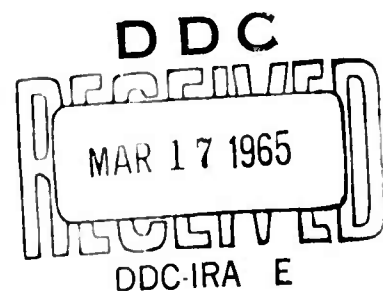
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FRICTION HYDRO PNEUMATIC SUSPENSION SYSTEM

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BACKGROUND

Mobility has been a key factor in many historic victories and defeats throughout the history of warfare. The defeat of France in 1940 can be attributed primarily to the rapid employment of German armored and mechanized forces that spearheaded the attack and achieved surprise. The excellent roadnets of Western Europe and favorable terrain enabled these units to move rapidly to their objectives and maintain an effective supply line. However, similarly equipped units were defeated decisively in Russia, where deep mud hindered tactical movement and adequate roads were either nonexistent or overloaded. Even the brilliant penetrations of General Patton's Third Army were dependent principally on the existence of good roadnets, and at the end of World War II, the mobility of armored and mechanized units still depended to a great extent on the availability and the condition of roads. During the post-war period, greater emphasis was placed on the achievement of a higher degree of ground mobility. This objective is particularly important when it is considered that there are vast land areas of the world where few roads exist and destruction resulting from the use of nuclear weapons may make the use of existing roads impossible.

Military requirements generated by the combat arms generally reflect the need for tactical vehicles that are capable of achieving good cross-country speeds over all types of terrain, as well as other mobility criteria such as swimming, deep-water fording, and air transportability. The U. S. Army Tank Automotive Center, through its research and engineering facilities, and in coordination with other research and development agencies, is working toward the solution of many technical problems which must be overcome in order to equip combat forces with vehicles which will provide a high degree of mobility under a wide variety of terrain conditions.

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One vehicle component that significantly limits the achievement of good cross-country speeds is the suspension system. Even today, with the many technological advances that have been made in the design of automotive components, the speed of a tank generally does not exceed 5-7 miles per hour over extremely rough terrain because of undesirable vehicle vibration characteristics at higher speeds.

Under the direction of the Components Research and Development Laboratories, U. S. Army Tank Automotive Center, an extensive engineering investigation of suspension systems was initiated in order to more clearly define the dynamic relationships between a vehicle and the terrain over which it must travel, and to establish parameters which can be applied directly to vehicle design toward achieving a significant improvement in vehicle cross-country mobility.

DESIGN PARAMETERS

Current research in the field of vehicle dynamics indicates that the ability of a vehicle to traverse rough terrain rapidly is dependent primarily on vehicle vibration characteristics which, in turn, are related to spring rate, damping, vertical wheel travel, and contour of the ground wave. There are other parameters such as angle of approach, angle of departure, weight distribution and track configuration which influence obstacle and soft-soil performance.

Effective operation of a tank is limited significantly by oscillation about the transverse axis (pitch axis) located directly below the center of gravity of the vehicle at the level of the wheel hub. Vehicle pitching, induced by terrain irregularities, imposes severe limitations on the ability of the crew to perform their duties effectively, and particularly limits the use of optical instruments for observation and target acquisition. Also, vehicle pitching interferes with efficient performance of other operations such as driving, and gun loading, and may result in personnel injuries and premature failure of vehicle components. Cross-country speeds, therefore, have been limited generally to 10 miles per hour or less in order to retain a reasonable degree of combat effectiveness. To remove or significantly diminish these limitations, it is necessary to obtain, through proper design and selection of components, better vehicle vibration characteristics where the pitch angle decreases as speed increases, and a level of vertical and angular acceleration is maintained as low as possible when traversing a given piece of terrain. This can be accomplished to some extent by decreasing the vehicle natural frequency, thereby decreasing the resonance speed over a given ground wave.

Because the natural frequency of pitch (f_0) is directly proportional to the spring rate (k) (Figure 1a), it is apparent that a reduction in the spring rate will result in a corresponding

reduction in the natural pitch frequency. Since a similar relationship exists between the natural frequency of bounce (f_y) and the spring rate (k) (Figure 1b), the natural frequency of bounce will be reduced also, if the spring rate is reduced. The selection of a soft spring has some limitations, however, and these limitations must be taken into account, or the advantages gained by lowering the natural frequency will be counteracted by the creation of conditions which are undesirable.

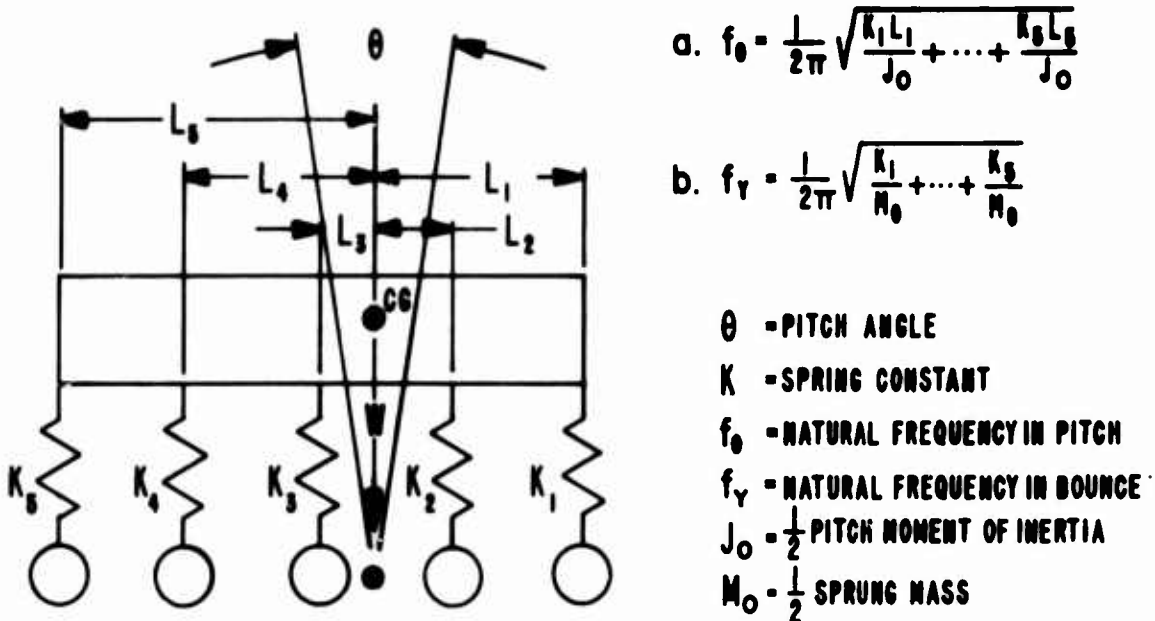


FIGURE 1 - SIMPLE VIBRATION SYSTEM

If a low spring rate is used without proper damping, a single impact, such as firing the main armament or passing a single obstacle, may cause prolonged oscillation. Although a soft spring will allow the wheels to move freely over minor surface irregularities without disturbing the vehicle hull significantly, it may not be capable of absorbing the kinetic energy transmitted by high-impact loads without bumpout. In order to achieve a significant measure of improvement in vibration characteristics, it is necessary to determine the optimum combination of springing and damping characteristics for maximum performance. Either a variable spring rate, variable damping, or both appear to be desirable in order to provide a suspension system sufficiently versatile so that its vibration characteristics can be optimized to suit a particular vehicle application and operational environment.

Analysis of vehicle ride dynamics indicated that a suspension system incorporating a low spring rate and a vertical wheel travel of approximately 16 inches would enable a 40-45 ton tank to traverse rough terrain at speeds of 24-27 miles per hour. Initial investigations were limited to the use of metallic springs which,

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when designed to provide the necessary characteristics, required an excessive amount of critical hull space. The use of external shock absorbers created additional hull drag which would degrade vehicle performance in deep mud.

The aim of ATAC's current development program was to provide an integrated, multiple function, compact, light-weight suspension system. The resulting design characteristics included internal damping, a low spring rate, variable ground clearance, suspension lockout and a 14-18 inch wheel travel. In order to accomplish these objectives, non-metallic springs were included in the analysis. As a result, the hydropneumatic spring emerged as the media which offered the greatest potential within the present state of the art because of its inherent high-energy storage capacity and nonlinear spring rate.

Since the hydropneumatic spring adheres to the Gas Laws and closely approximates an adiabatic process ($PV^\gamma = \text{Constant}$), gas pressure increases exponentially as increasing wheel forces are transmitted to the spring:

$$(1) \quad P_2 = P_1 \left[\frac{V_1}{V_2} \right]^\gamma$$

Wheel deflection can be related directly to a change in gas volume by use of a linkage between the suspension arm and the spring; therefore, the spring rate (k) is proportional to the static pressure (P_1) and the change in gas volume (ΔV).

$$(2) \quad k \approx P_1 \left[\frac{V_1}{V_2} \right]^\gamma \quad \text{OR} \quad k \approx P_1 \left[\frac{V_1}{V_1 - \Delta V} \right]^\gamma$$

In order to transmit wheel forces to the spring and provide a means of damping, a three-vaned hydraulic rotary actuator offered several advantages. The torque capacity of a vane-type rotary actuator is proportional to the total vane area, the mean radius of the vanes, the number of vanes and the hydraulic pressure.

$$(3) \quad T = P A R_m N$$

A three-vaned actuator provides high torque capacity in a compact unit without developing excessive hydraulic pressures. Further increase in the number of vanes would not provide sufficient angular

deflection, and consequently would limit vertical wheel travel. The use of a hydraulic rotary actuator enables the spring characteristics to be varied by rotating the roadarms, thereby varying the vehicle ground clearance and the static pressure in the spring.

DEVELOPMENT

The development of a tank suspension system based upon the rotary hydropneumatic machinery was initiated in 1960. The design began with an 8100 pound static wheel load and dynamic loads of over 100,000 pounds. Four automatically controlled operating heights were to be provided, and a vertical wheel travel of 14-18 inches was required, using a 16-inch roadarm and 33-inch diameter roadwheels.

Allowing the necessary space for porting and for the physical size of the vanes, it was determined that 73° of rotation could be obtained from a three-vaned hydraulic rotary actuator without physical contact between movable and fixed vanes. An angular displacement of 73° resulted in a total vertical wheel travel of 18.4 inches (Figure 2). Four operating positions were selected within the 73° limit in order to provide some variation of wheel travel and vehicle ground clearance. Neglecting friction, and

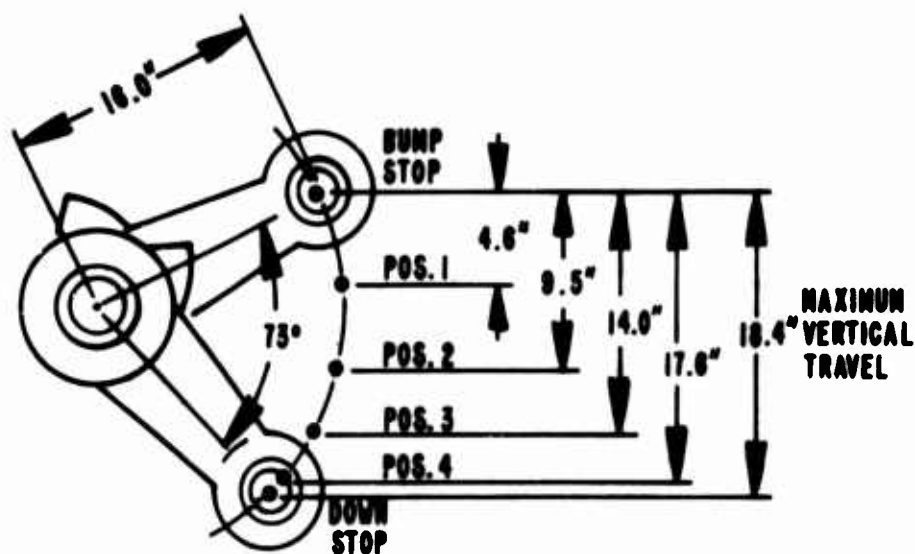


FIGURE 2 - VERTICAL WHEEL TRAVEL

using a rotary actuator 9 inches in diameter, calculations revealed that a torque of 64,000 lb.-in./psi. could be developed, and that a total of 81.5 cu. in. of oil would be displaced if the total angular

displacement of 73° were used. Assuming a 95% mechanical efficiency for the rotary actuator, the maximum torque required to lift the vehicle occurs in position number 1 (Figure 2) where the roadarm is almost horizontal and creates the longest lever arm. In this position 135,800 lb.-in. are required to lift the vehicle, resulting in a requirement for a 2110 psi hydraulic pressure. Position number 1 can be maintained with only 1915 psi, since friction helps to support the vehicle. A 3000 psi pressure-compensated hydraulic pump with a maximum flow of 28.5 gallons per minute at full flow was selected to supply the system. Since the maximum oil capacity of a single actuator is 81.5 cu. in., and the maximum lift pressure is 2110 psi, the pump is capable of raising the vehicle rapidly. To raise the vehicle from the upstop to position number 4 (69°) requires 77 cu. in. of oil per actuator. The necessary oil can be supplied by the pump in less than 7 seconds, with the pump running at 2700 rpm.

Design of the hydropneumatic spring was governed by several considerations:

1. The oil capacity required is 81.5 cu. in.
2. The initial gas precharge should be a minimum.
3. Maximum dynamic pressure should be limited to about 6000 psi.
4. The free piston type accumulator should, if possible, fit inside the arm shaft in order that the complete suspension unit might be constructed as a package.
5. Adiabatic gas compression would be assumed.

Calculations were made for all wheel positions using the following relationships and using a maximum gas volume of 132 cu. in. and assumed precharges (Figure 3):

$$(4) \quad P_3 = P_2 \left[\frac{P_0 V_0}{P_0 V_0 - P_2 \Delta V_u} \right]^{1.4} \quad \text{WHICH IS DERIVED FROM THE RELATION: } P_3 = P_2 \left[\frac{V_2}{V_3} \right]^{1.4}$$

By calculation, computer simulation and laboratory test it was determined that a gas precharge of 1350 psi would be suitable, since it resulted in a maximum pressure of 6620 psi, which is close to the specifications established. Lower peak pressures can be obtained but higher precharges would be required. Using the data already

established, the spring rate curves for each operating position were calculated and plotted (Figure 4).

Damping in rebound is achieved by passing oil through a fixed orifice damping valve. The valve chosen was selected primarily from experience gained in other hydraulic applications, and its adequacy is to be determined through tests of the complete system. Calculated spring rates in the vicinity of the static position were on the order of 350-900 lb/in, depending upon the position of the roadarm, and calculated vehicle natural frequencies were on the order of 39-62 cycles per minute.

The components described above were fabricated, subjected to laboratory tests and installed in a test vehicle. Early prototypes used external accumulators, but redesign in early stages of development resulted in the configuration shown in Figure 5. The need for a stable firing platform for the main armament and a means of preventing immobilization of the vehicle due to hydraulic failure precipitated the development of a friction brake system. This system provides an additional capacity for absorbing great dynamic loads imposed on front and rear wheels when traversing rough terrain at high speeds.

The first redesign resulted in the addition of friction disc brake units to front and rear suspension units as shown in Figure 6. During the final 7° of upward travel of the roadarm, a cam on the arm shaft causes the brake piston to displace approximately 3 cu. cm. of oil into an annular cavity behind a ring-type piston, thereby raising the oil pressure to approximately 1400 psi and engaging the friction discs. The brake unit is capable of absorbing 700,000 lb-in of torque, thereby reducing the kinetic energy transmitted to the hull. Approximately one half of the bumpout energy is absorbed in the brake unit, and the remainder is divided between the hydro-pneumatic spring and the solid metal bump stop. An important function of the brake unit is to provide a means of locking the suspension in a selected position to provide a rigid firing platform for the vehicle weapons system. Actuation in this mode is accomplished by applying oil pressure directly to the brake piston by means of an external control valve. Pressure is maintained by an accumulator during periods in which the hydraulic pump is inoperative.

Oil entering or leaving the hydro-pneumatic spring passes through a fixed orifice damping valve which is designed to provide damping in rebound only. Wheel movement induced by terrain roughness results in the transfer of oil from the actuator to the hydro-pneumatic spring (through the damping valve in which the poppet is unseated) allowing relatively unrestricted flow. In rebound, the poppet is seated and oil returning to the actuator is forced through a series of small orifices to provide the necessary damping.

EVALUATION AND TESTING

Computer simulations conducted at the U. S. Army Tank Automotive Center enabled a preliminary technical evaluation to be made prior to actual on-vehicle tests. Using an analog computer and terrain traces of the Perryman Cross-country Course at Aberdeen Proving Ground, a constant-speed evaluation was made of two equally-weighted vehicles. One vehicle was equipped with the Friction Hydro Pneumatic Suspension and the other with the current production torsion bar suspension. The results confirm the fact that a significant reduction in pitch and bounce vibration can be expected at speeds of 25 miles per hour using the Friction Hydro Pneumatic Suspension. A sample of the results is shown in Figure 7. Since it was necessary to make numerous assumptions in order to program the computer, it was considered essential to determine the suspension characteristics by on-vehicle testing.

In order that test conditions could be repeated, a test course was constructed using a prepared roadway and rigid obstacles which varied in height from 6 inches to 12 inches. Obstacles were spaced 20 feet apart, since this was considered to be the worst condition for the vehicles being evaluated. Two tanks sprung with torsion bars and one tank sprung with the Friction Hydro Pneumatic Suspension were used. The objective of the evaluation was to determine the maximum speed that each vehicle could achieve over a series of 10 obstacles of a given height, spaced 20 feet apart. Instrumentation was installed in order to record vertical and pitch accelerations, pitch angles and vehicle speed. Only experienced drivers were used and the limit of each vehicle was determined by the ability of the driver to maintain control of the vehicle. The three test vehicles were run over courses consisting of 6-inch, 8-inch and 12-inch obstacles in speed increments of 5 miles per hour.

In order to provide some criteria against which the vibration characteristics could be measured, it was necessary to establish a level of pitch amplitude and vertical acceleration which would be considered acceptable for effective operation of the vehicle. A review of the research in the field of vehicle dynamics conducted by Lehr in Germany⁽¹⁾ revealed that vertical acceleration in excess of 0.4g (12.9 ft/sec²) would interfere with efficient use of optical instruments of the tank's fire control system and that pitch amplitude should be reduced to approximately $\pm 2^\circ$ using the proper arrangement of springing and damping. The criteria of 0.4g (12.9 ft/sec²) falls within the "discomfort zone" (10-14 ft/sec²).⁽²⁾

The criteria established for this suspension evaluation was established as 10 ft/sec² or less for effective tank operation since it coincides very closely to the criteria established by Lehr and is also the lower limit of discomfort. A total pitch angle of 4° was arbitrarily selected as an acceptable limit, however, it is anticipated that this criteria will be more accurately determined by

a more detailed analysis of the human factors involved, as well as the physical capabilities of the optical instruments used in the tank.

Although evaluation of the Friction Hydro Pneumatic Suspension System has not been completed, the results achieved to date are significant. Maximum speed of the control vehicles sprung by standard production torsion bars was found to be limited to approximately 11 miles per hour over a course consisting of 10 six-inch obstacles, spaced 20 feet apart. At this speed pitching was so violent that drivers were unable to maintain control of the vehicles and experienced difficulty in remaining seated even with the aid of seat belts. The vehicle sprung with the Friction Hydro Pneumatic Suspension System negotiated the course at the maximum speed that could be achieved by its power plant, 22 miles per hour, without causing the driver to experience difficulty in steering or control. Testing was repeated using 8-inch and 12-inch obstacles. Although maximum speeds were reduced as the obstacle size was increased, the differential in speed capability remained approximately the same (Figure 8), even though the gross power-to-weight ratio of one of the control vehicles was 50% greater than the Friction Hydro Pneumatic test vehicle. The second control vehicle had an equal power-to-weight ratio. An advantage in power-to-weight ratio did not seem to affect the results significantly.

Analysis of the data recorded revealed that over the six-inch obstacles, pitch angles remained at a low level ($\leq 3^\circ$) on the Hydro Pneumatic test vehicle throughout the speed range (0-22 miles per hour). Pitch acceleration also remained at a low level ($\leq 5 \text{ rad/sec}^2$). The best performance achieved by the control vehicles resulted in pitch angles and pitch accelerations so high (16.6° and 11 rad/sec^2) that the driver was unable to pass the resonance speed (Figure 9). Vertical accelerations recorded show the same general characteristic (Figure 10). If the criteria for effective operation of the vehicle is established at 12.9 ft/sec^2 , as suggested by Lehr, the Friction Hydro Pneumatic test vehicle remains effective throughout the speed range (0-22 mph) while the control vehicles become increasingly ineffective at approximately 10 miles per hour.

It is difficult to measure a vehicle's total effectiveness, however, because of the many human and mechanical variables involved. In order to present an objective evaluation, no definite cutoff is shown in Figure 10 and only a gradual decrease in effectiveness is shown for accelerations above 10 ft/sec^2 . Vibrations increased in magnitude as the size of the obstacles was increased, but in all cases the Friction Hydro Pneumatic test vehicle was capable of negotiating the course at higher speeds than the control vehicles.

The evaluation is continuing in order to determine the performance limits when wave length and amplitude of the ground wave

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are not constant. Data collected during this evaluation will be applied to the development of other vehicle suspension systems where high cross-country speed is a required characteristic.

CONCLUSIONS

1. It is possible to design a suspension system which provides a 41-ton tank with approximately 100% increase in cross-country speed without increasing the power-to-weight ratio of the vehicle.

2. A suspension system incorporating a low spring rate and adequate damping will provide a significant reduction in pitch and bounce vibrations, thereby increasing vehicle cross-country speed capability without degrading the effectiveness of the crew or the vehicle weapons system.

3. It is feasible to design cross-country vehicles with a speed capability of 30-35 miles per hour within the present state of the art.

4. A major advancement in mobility machinery has been achieved by means of a theoretical analysis of suspension design characteristics and performance parameters.

An extensive amount of research and engineering work must still be accomplished in order to develop suspension systems capable of providing efficient operation at high speeds over all types of terrain, particularly in the fields of springing and damping media, suspension geometry, terrain characteristics and terrain sensing devices.

The promising results achieved to date have already had a significant impact on the design of suspension systems for military vehicles. Perhaps even more important is the impetus given to the initiation of companion research and development programs which will contribute toward improving the effectiveness of future military vehicles under a wide variety of terrain conditions.

REFERENCES

- (1) Lehr, E. "The Springing and Damping of the Suspension of Armored Vehicles", Berlin, January 28, 1944 (Translation of presentation lecture to the Ministry for Armaments in War Production).
- (2) Ordnance Proof Manual, Volume II, Aberdeen Proving Ground, Maryland, OPM 60-305, "Human Engineering", 20 November 1957, Page 5.

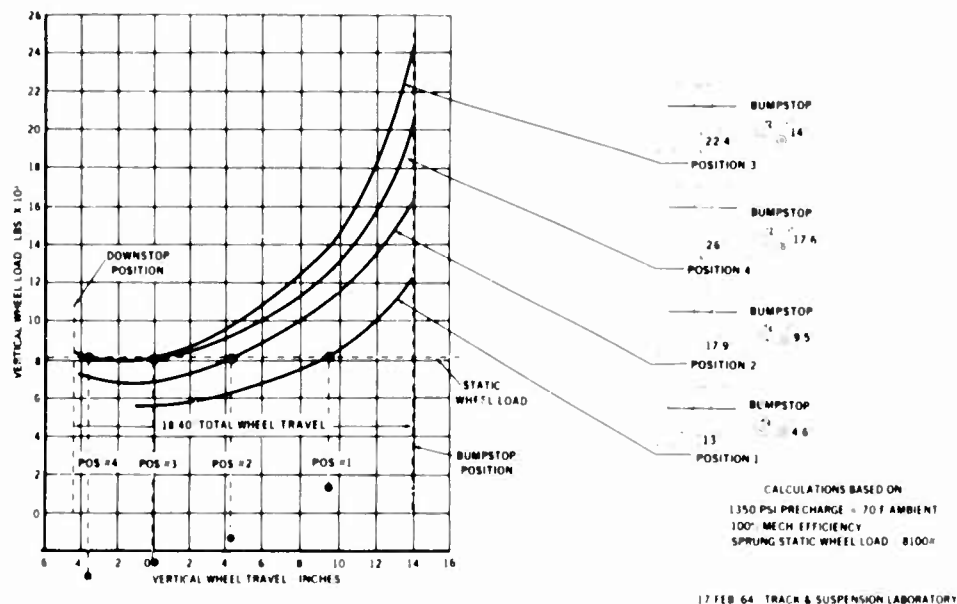


FIGURE 3 CALCULATED SPRING CHARACTERISTICS

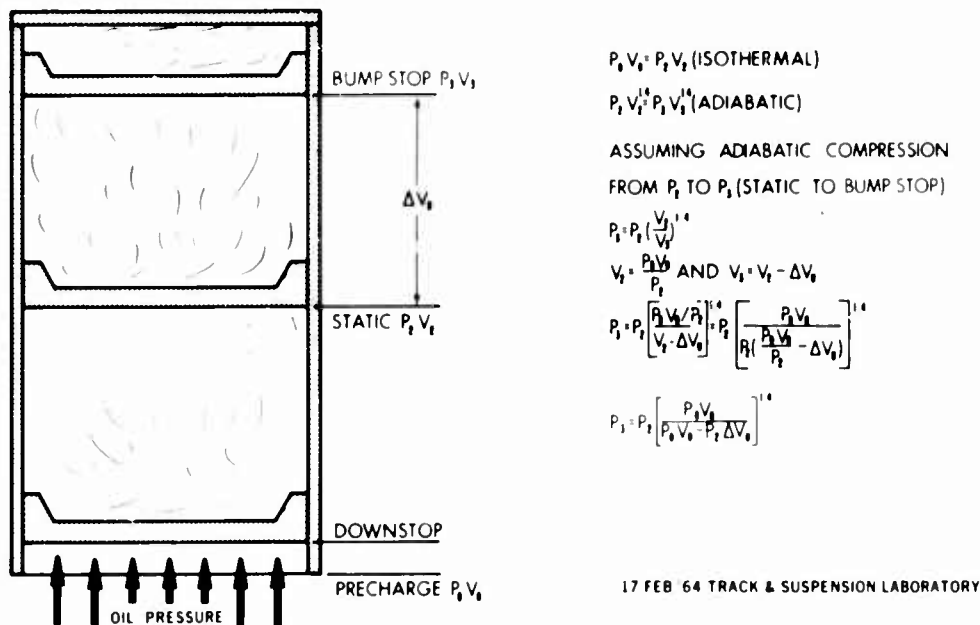
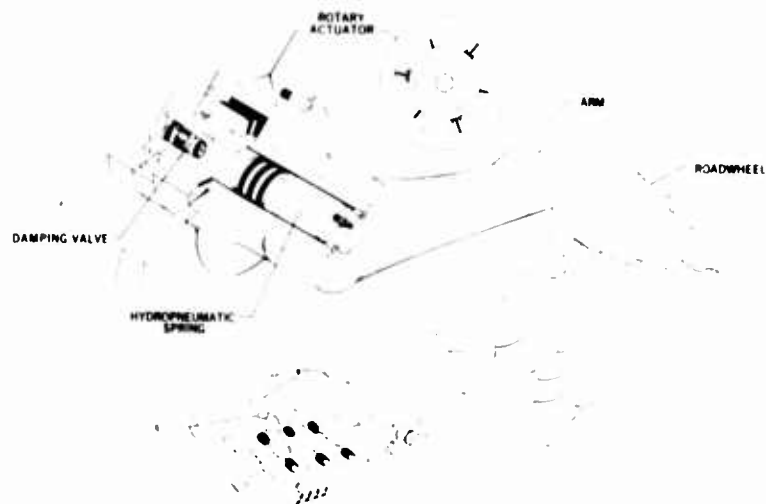
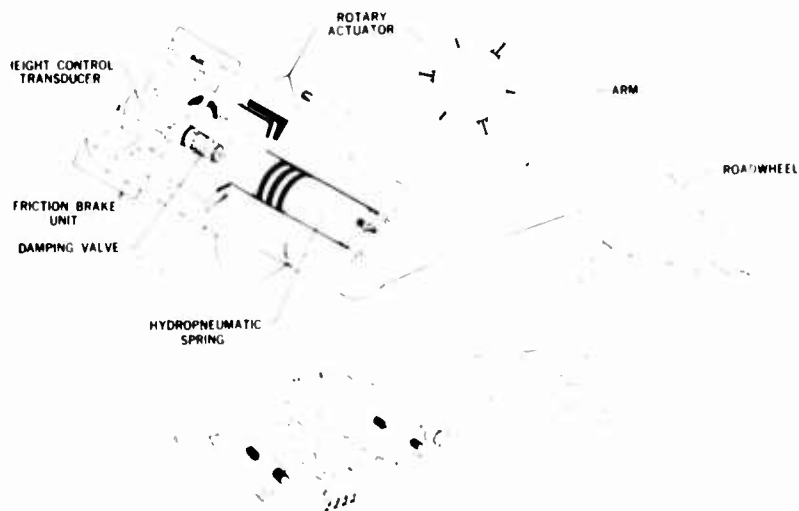


FIGURE 4 SPRING CALCULATIONS



17 FEB 64 TRACK & SUSPENSION LABORATORY

FIGURE 5 INTERMEDIATE UNIT



17 FEB 64 TRACK & SUSPENSION LABORATORY

FIGURE 6 FRONT & REAR UNIT

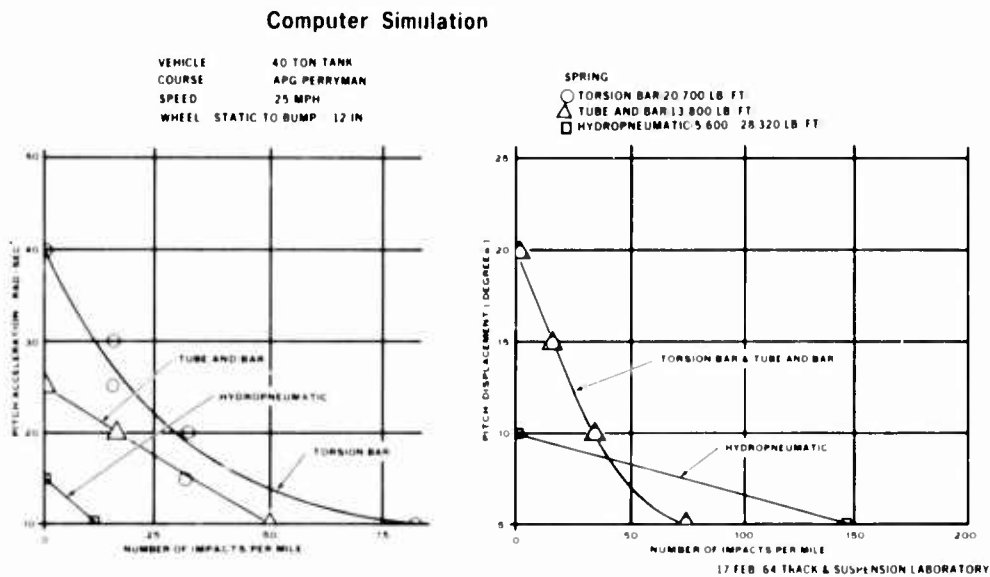


FIGURE 7 WHEEL TRAVEL RESEARCH STUDY

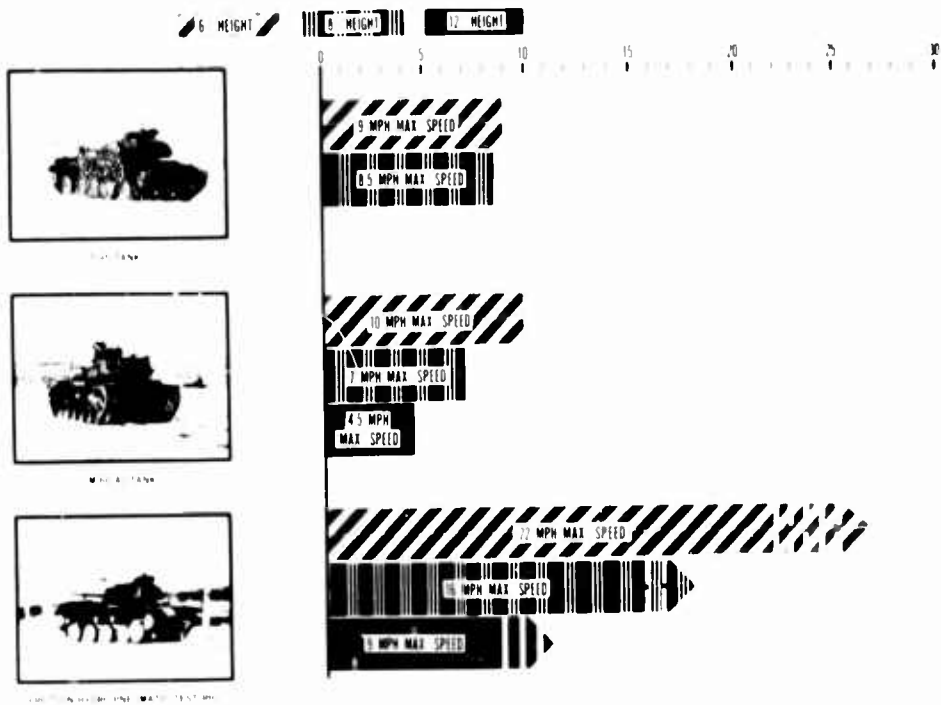


FIGURE 8 MAXIMUM SAFE SPEED OVER SERIES OF TEN OBSTACLES SPACED 20 FEET APART

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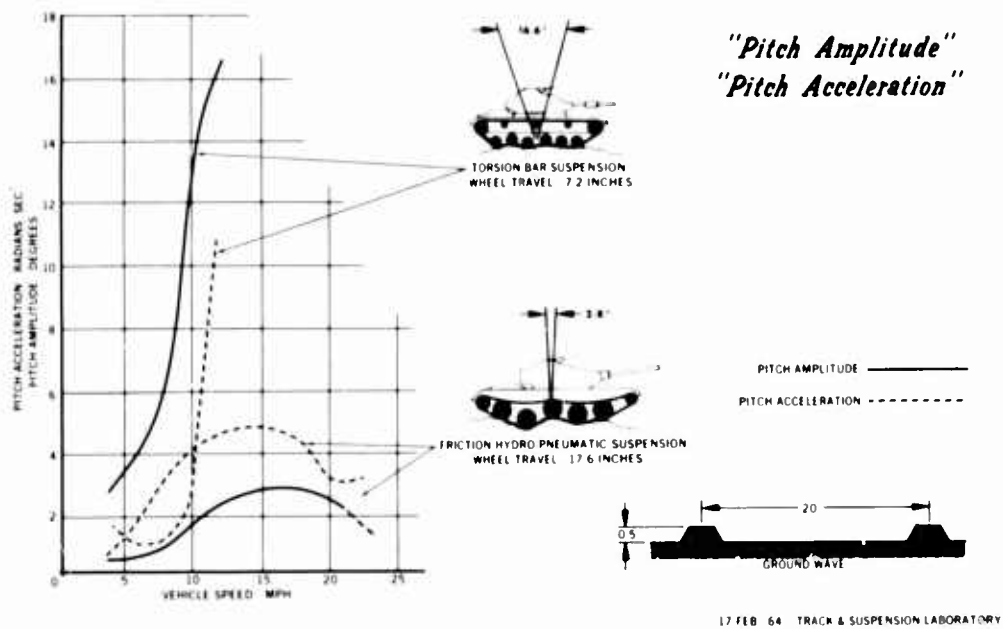


FIGURE 9 VIBRATION CHARACTERISTICS

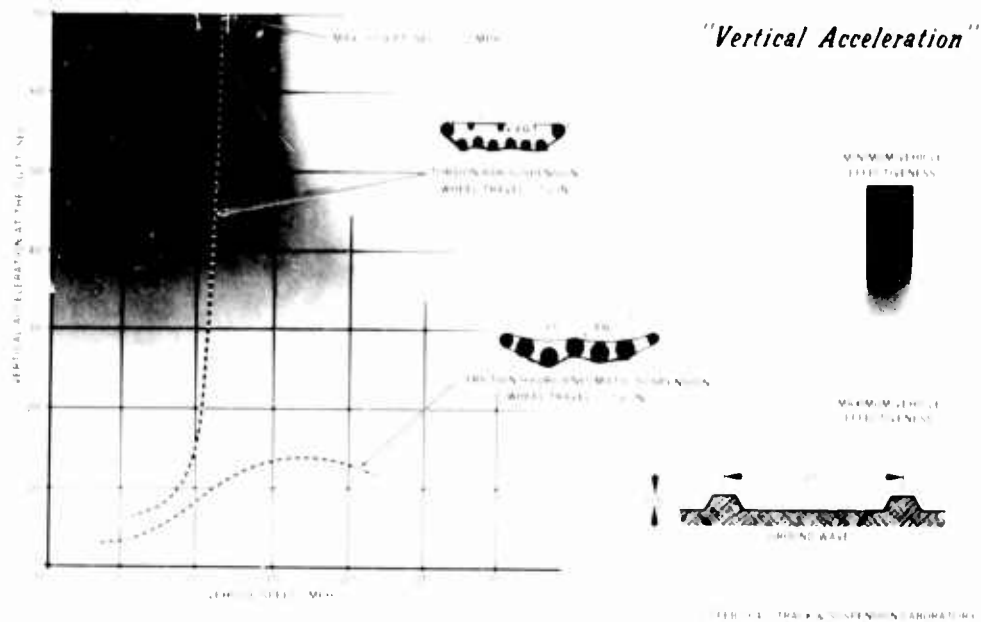


FIGURE 10 VIBRATION CHARACTERISTICS